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Lafreniere et al.

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(54) **DRIVING PULLEY FOR A CONTINUOUSLY VARIABLE TRANSMISSION**

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PCT Pub. Date: **Mar. 24, 2011**

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(Continued)

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CPC ***F16H 63/065*** (2013.01); ***F16H 55/56*** (2013.01); *F16H 61/66259* (2013.01); *F16H 61/66272* (2013.01)

(58) **Field of Classification Search**
CPC F16H 59/00; F16H 61/00; F16H 63/00
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See application file for complete search history.

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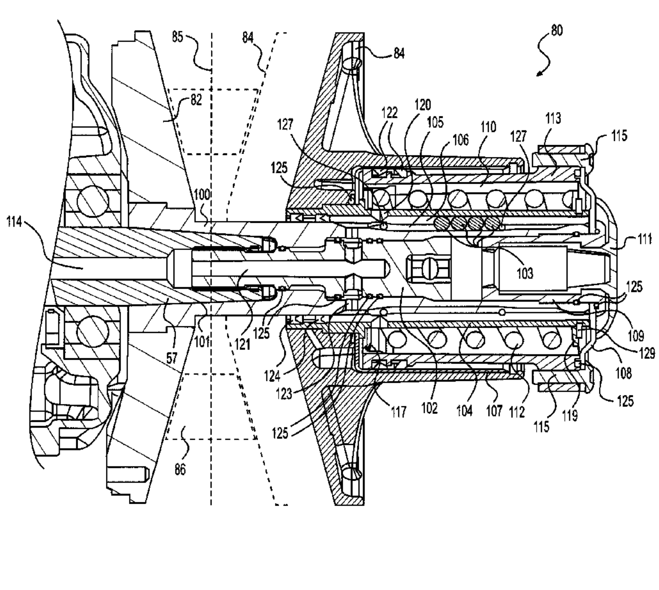
(57) **ABSTRACT**

A driving pulley for a CVT has a shaft, a fixed sheave mounted on the shaft, a first sleeve disposed around the shaft and being operatively connected to the shaft, a movable sheave mounted on the first sleeve, a second sleeve disposed around the first sleeve and being connected to the shaft, a spring biasing the movable sheave away from the fixed sheave, and a CVT chamber having an annular cross-section. The fixed and movable sheaves are adapted to receive a belt therebetween. The CVT chamber has at least one opening adapted for fluidly communicating the CVT chamber with a hydraulic fluid reservoir. Hydraulic pressure in the CVT chamber biases the movable sheave toward the fixed sheave. The CVT chamber has an inner wall formed by the first sleeve, an outer wall formed by the second sleeve, an outer end, and an inner end formed by the movable sheave.

12 Claims, 14 Drawing Sheets



(51) **Int. Cl.**

F16H 9/14	(2006.01)
F16H 63/06	(2006.01)

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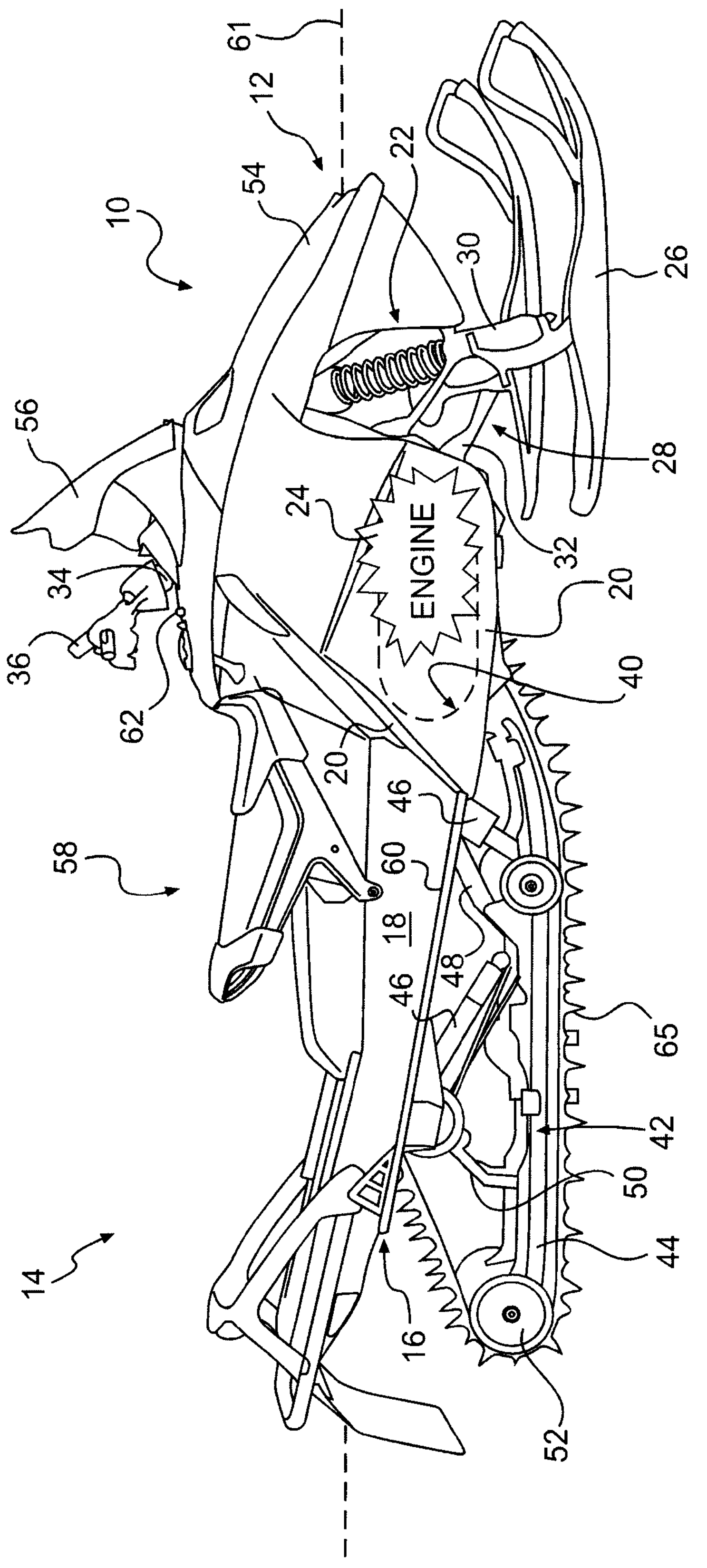
ENGINE
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FIG. 1

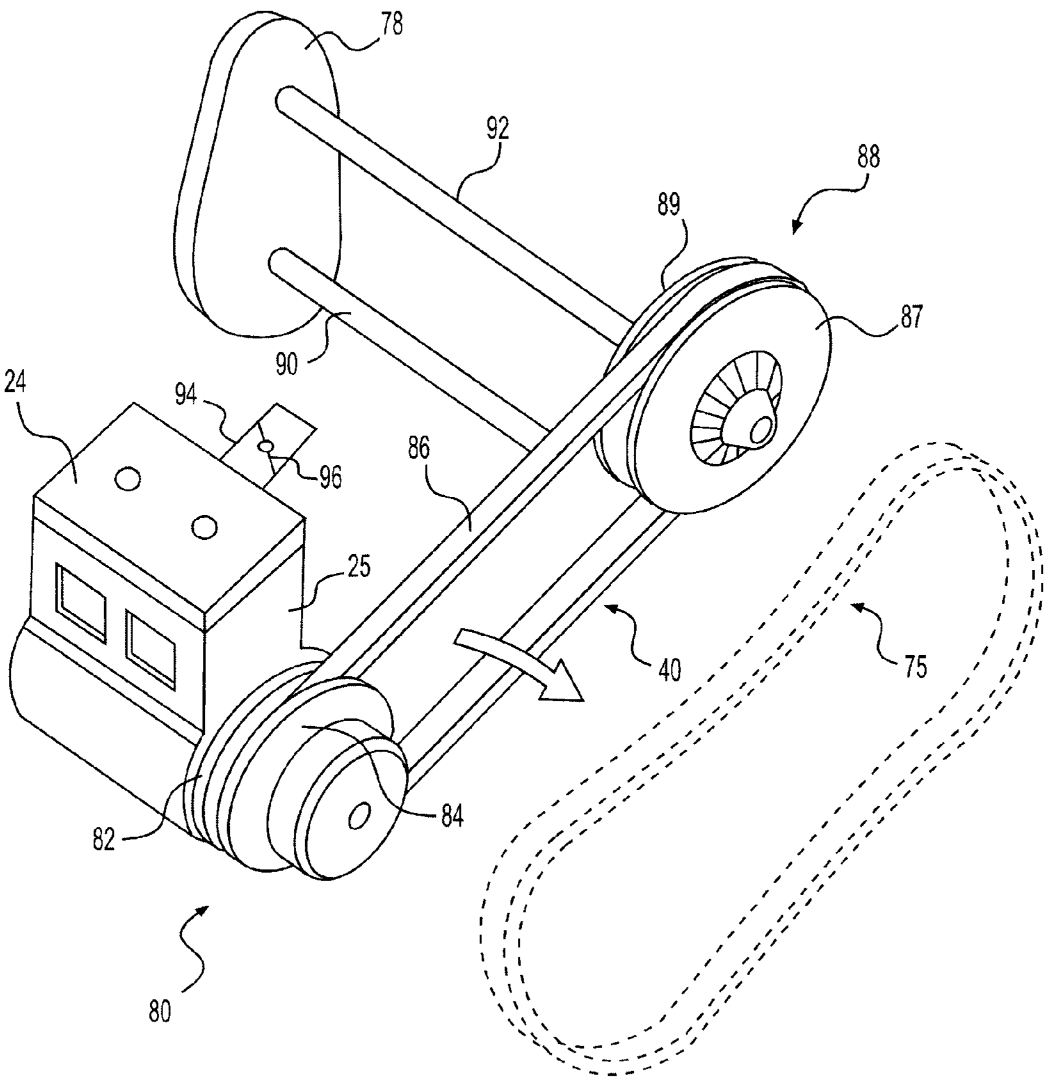
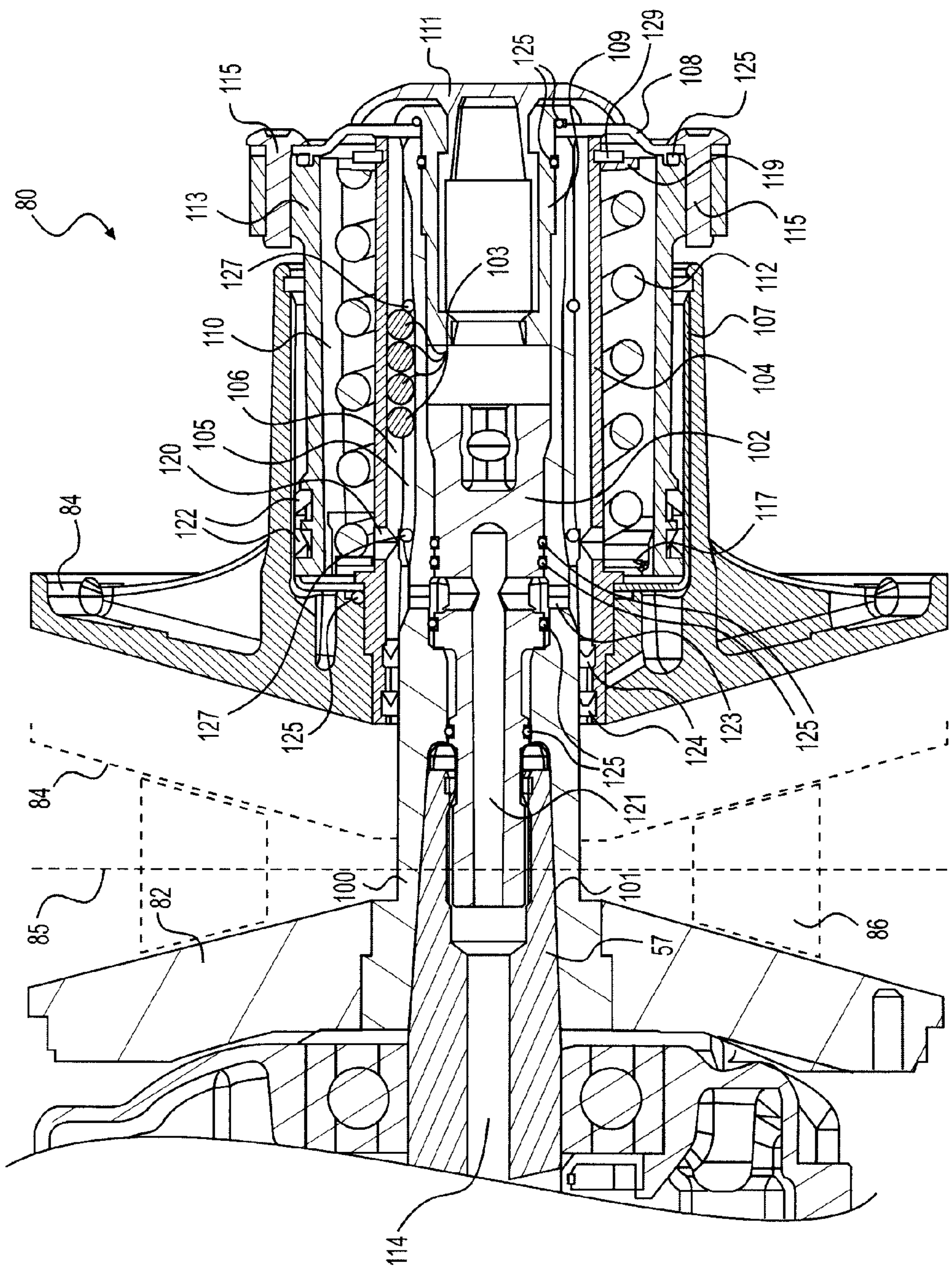


FIG. 2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101
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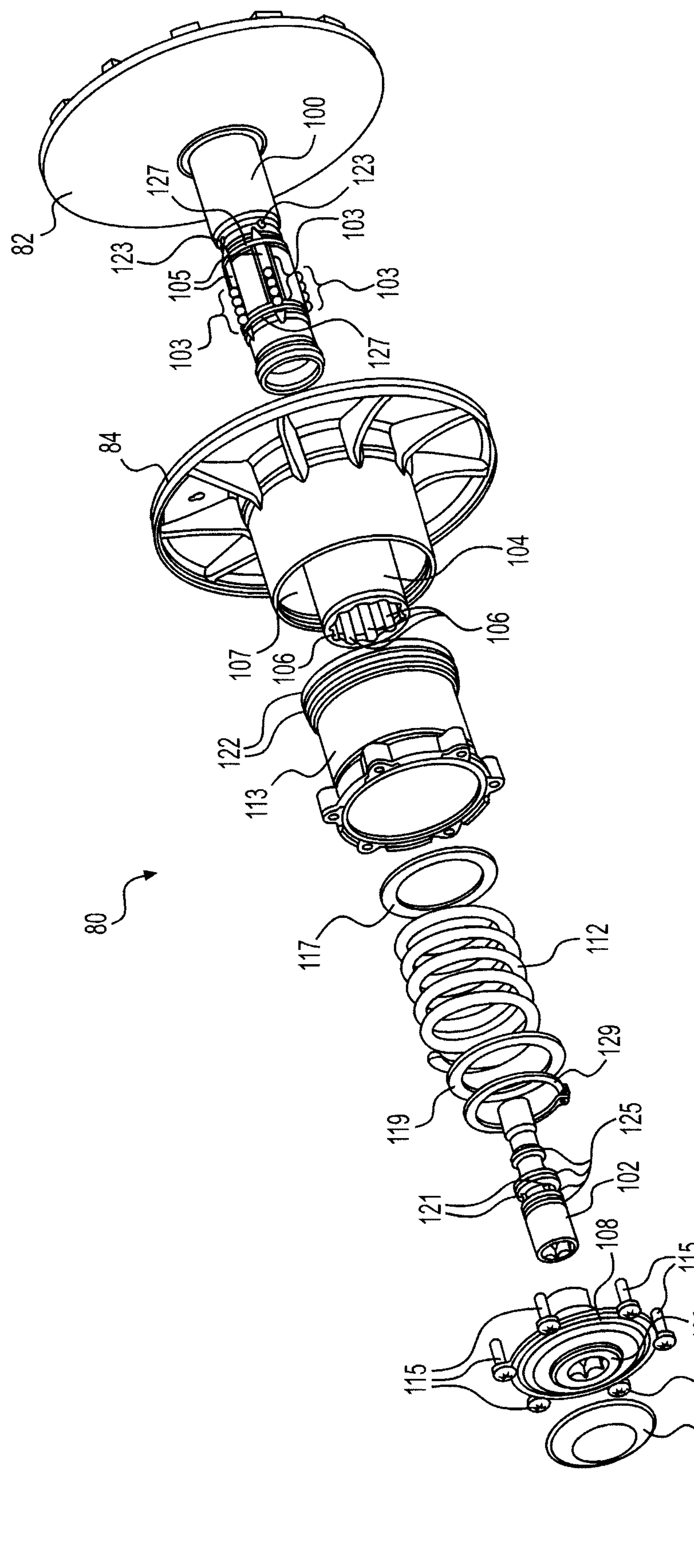
86
114

FIG. 3A



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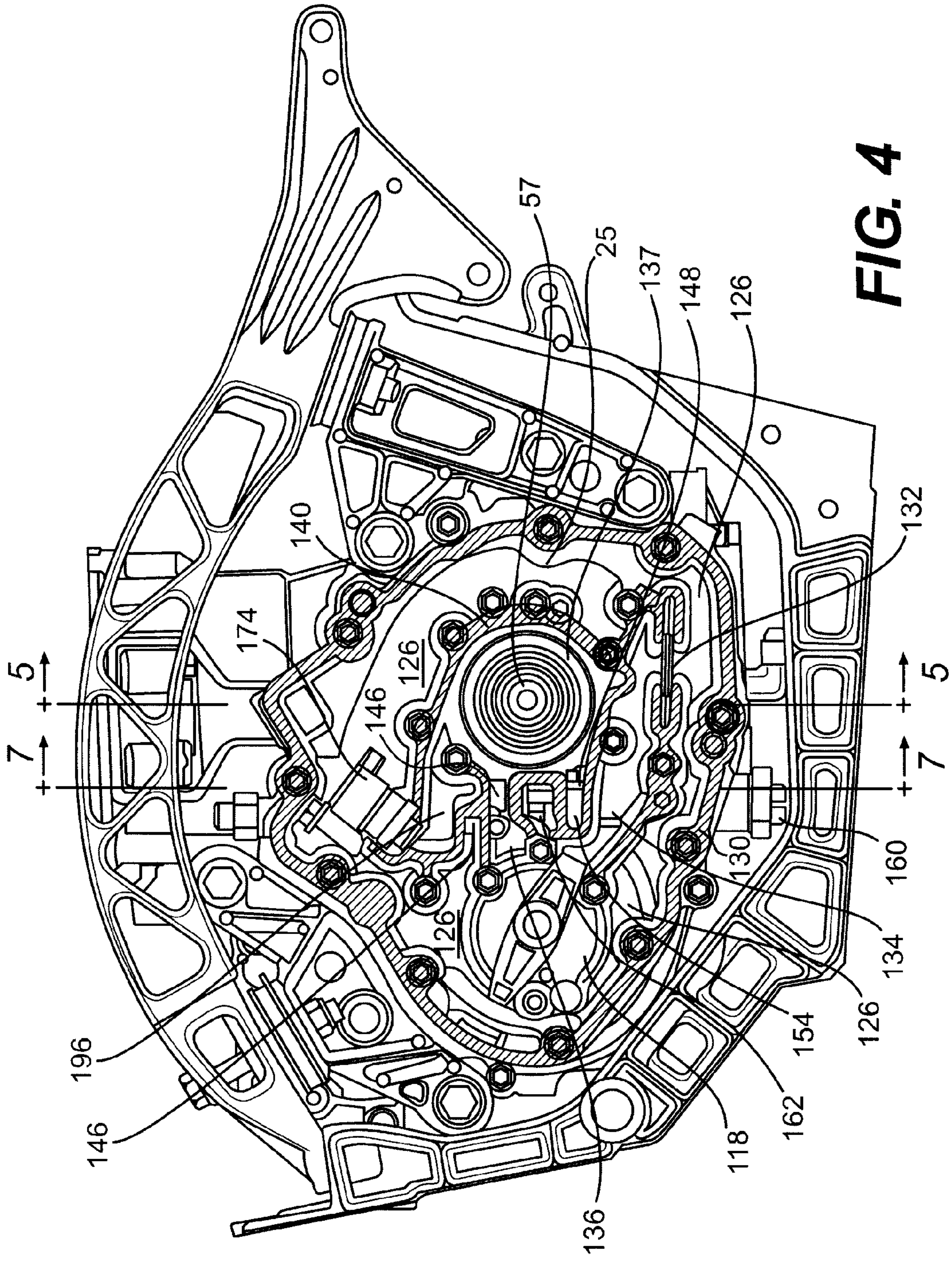
111

FIG. 3B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132
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136
196
146

FIG. 4

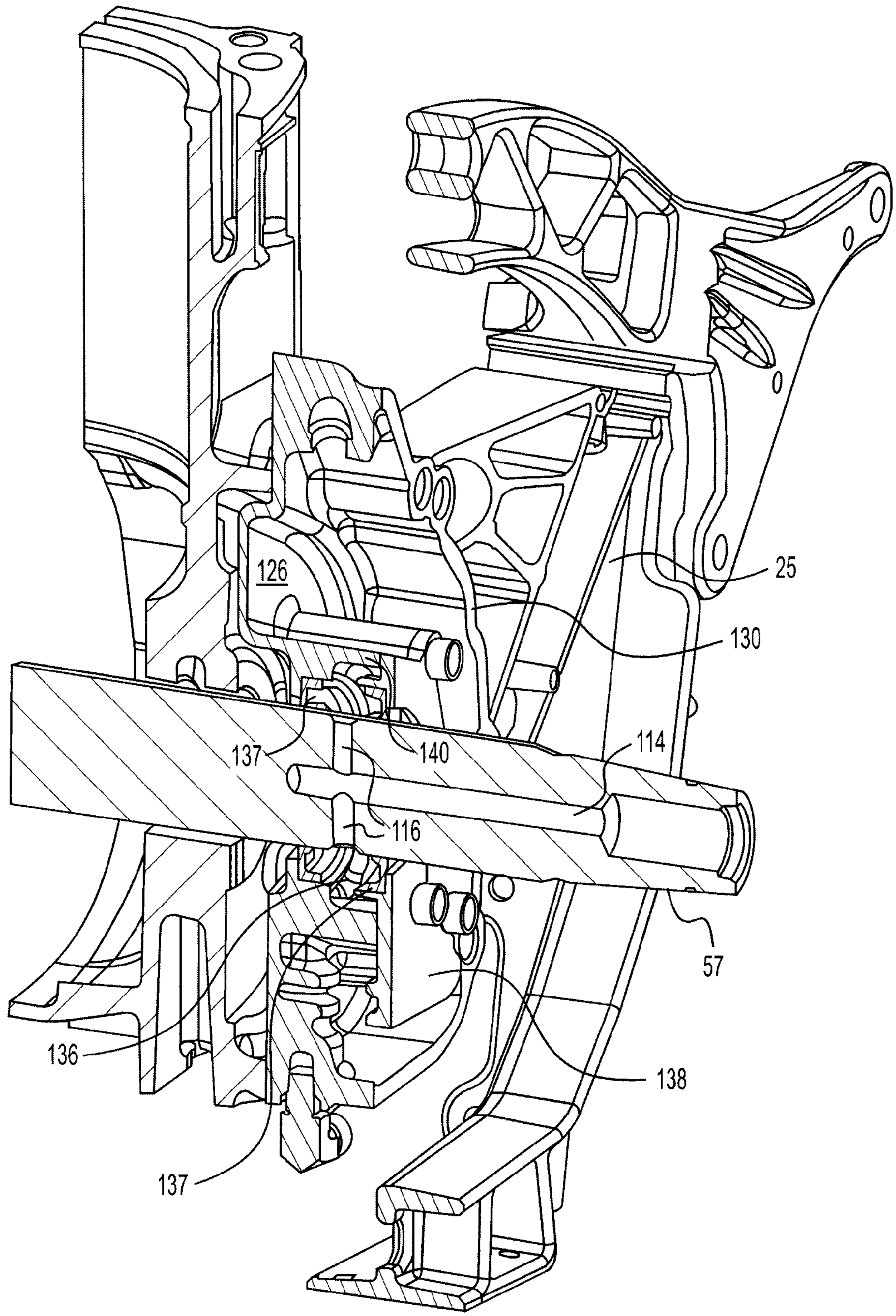
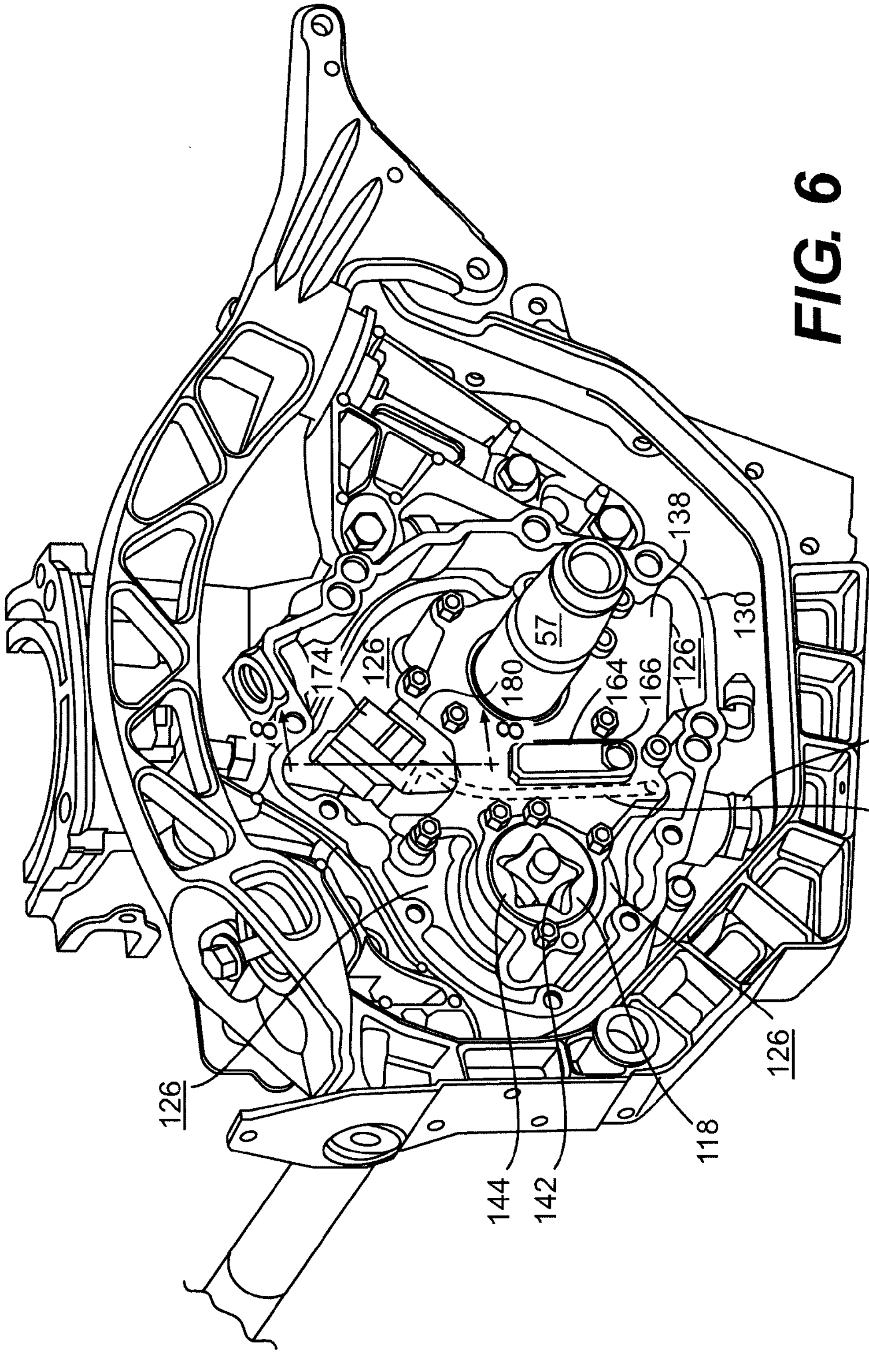


FIG. 5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168
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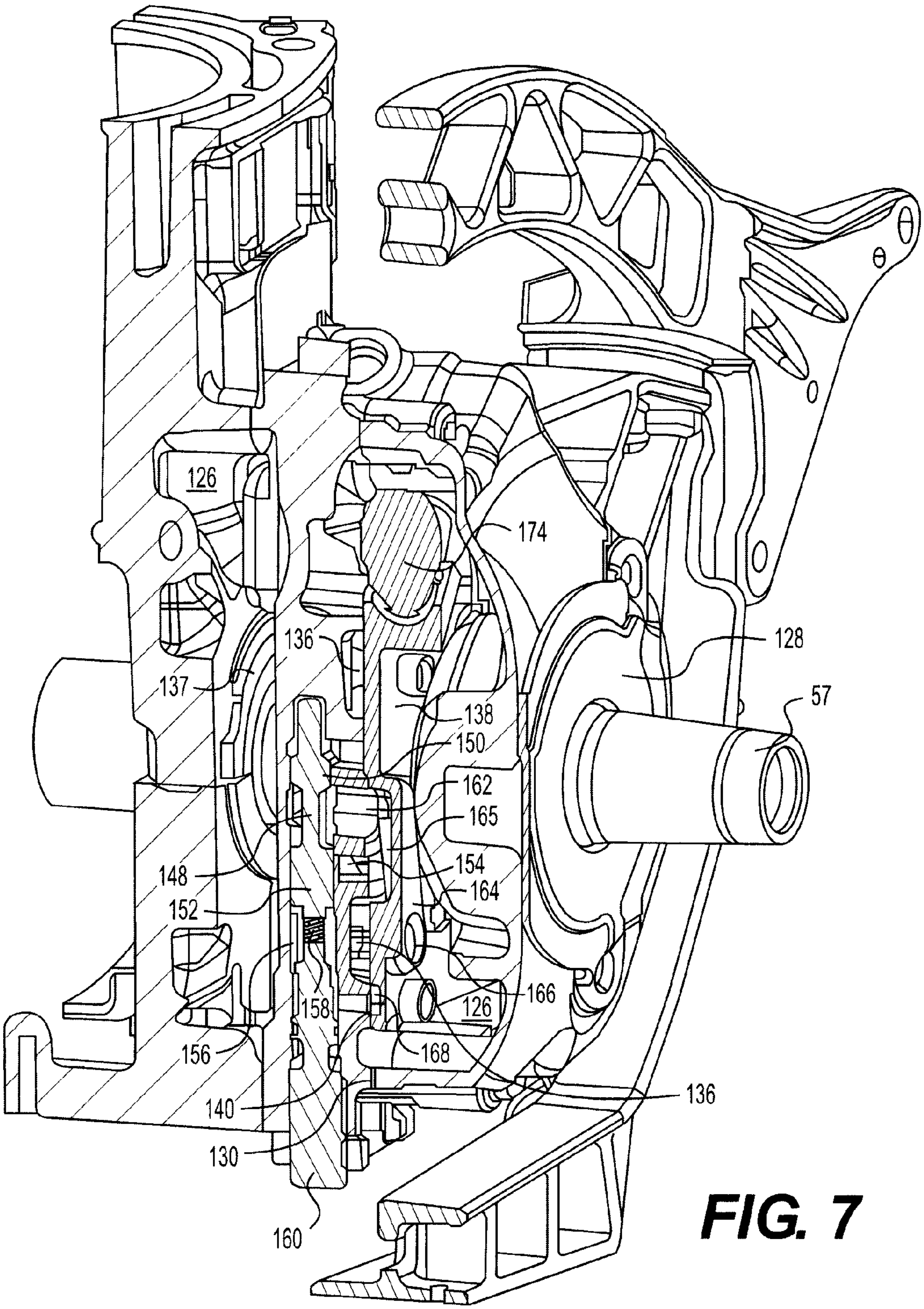
126
144
142
118

FIG. 6



126
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126
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140
136
130
160

FIG. 7

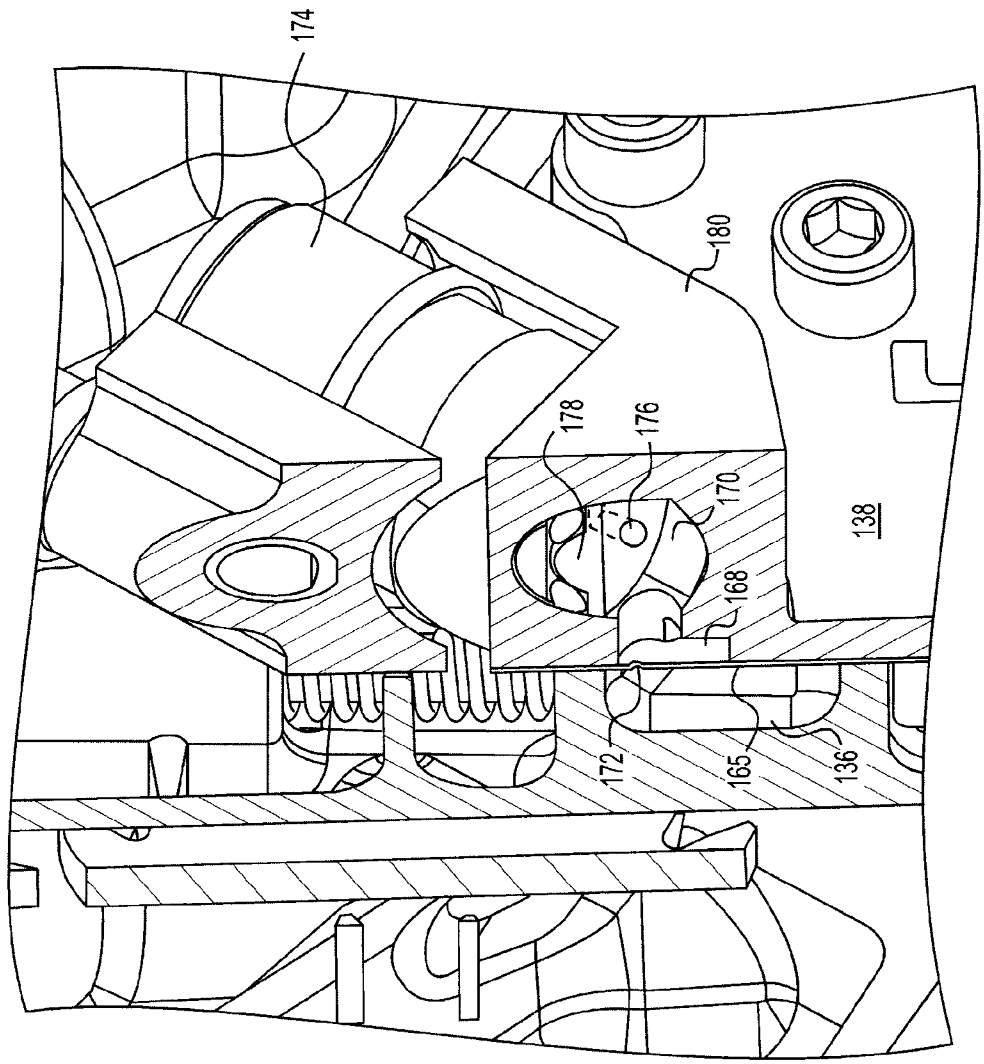

FIG. 8

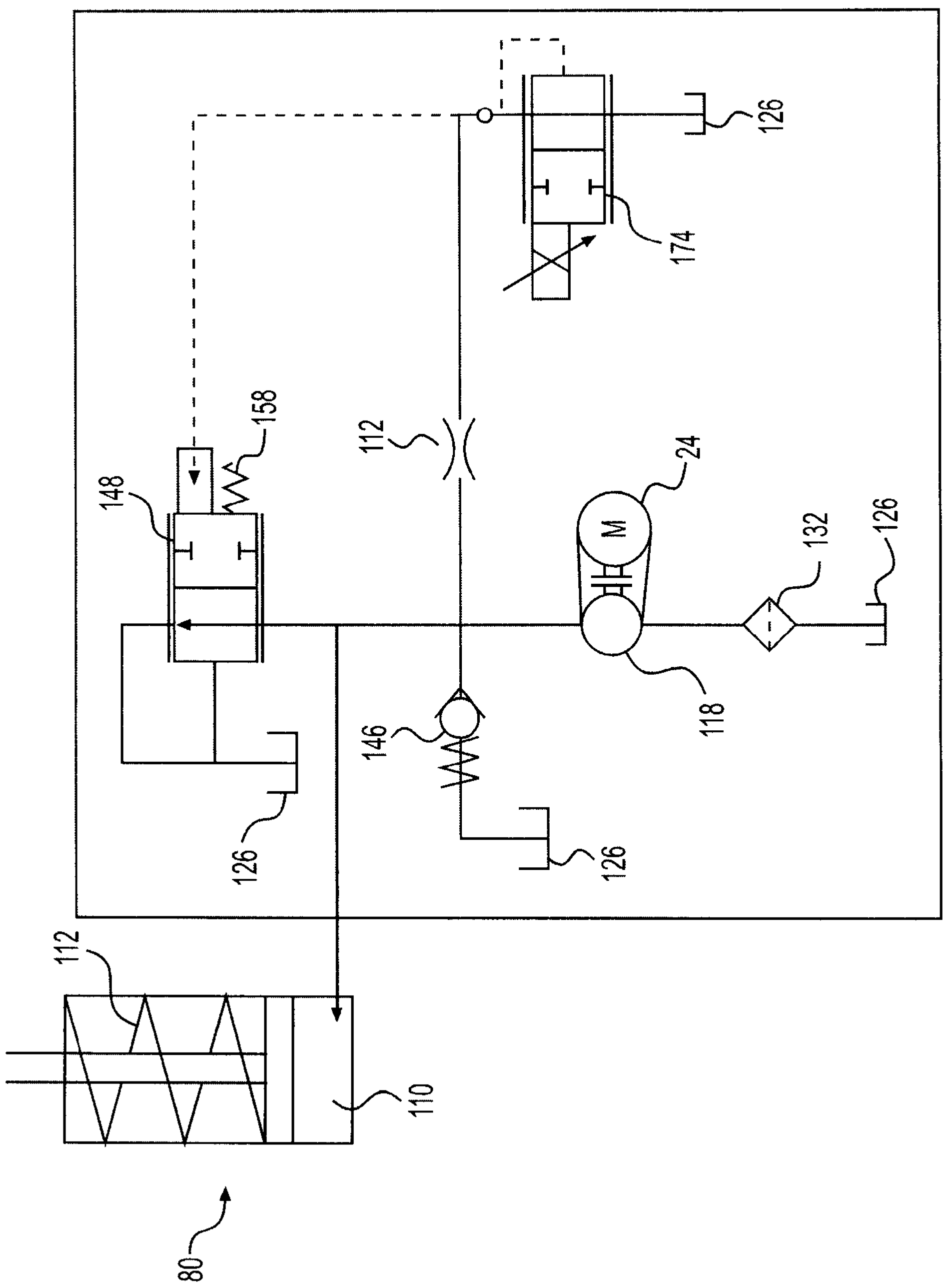

FIG. 9

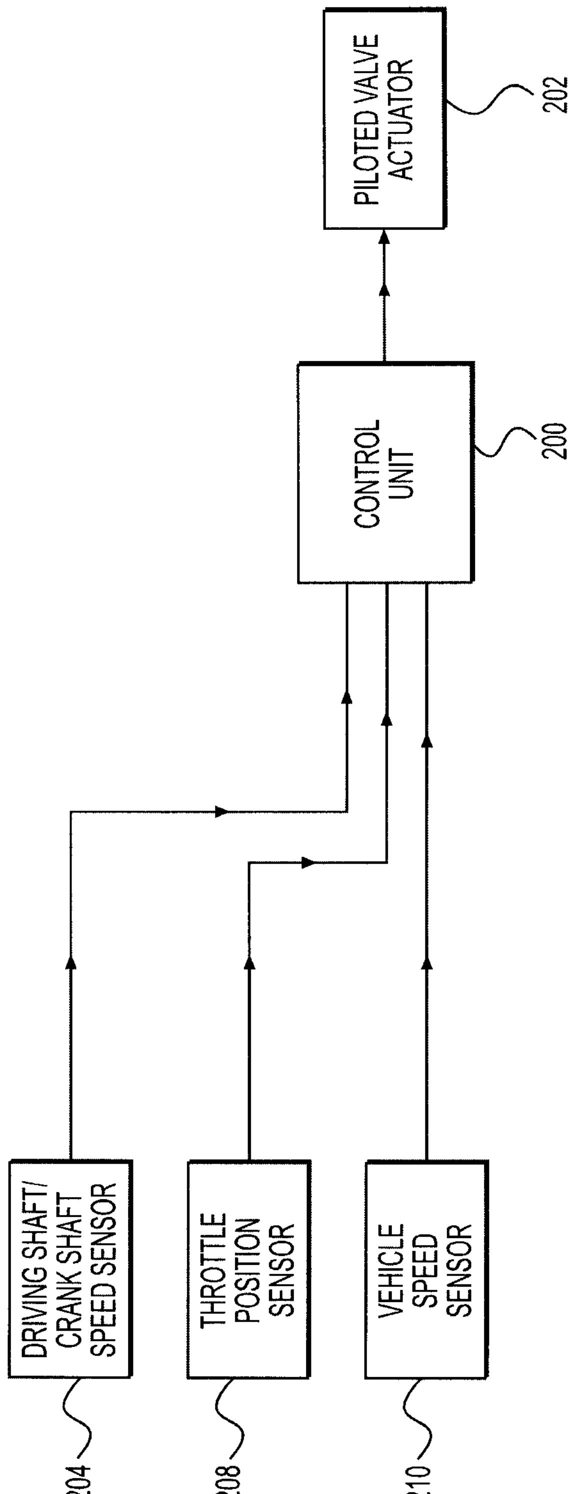
PILOTED VALVE ACTUATOR
202
CONTROL UNIT
200
DRIVING SHAFT/ CRANK SHAFT SPEED SENSOR
204
THROTTLE POSITION SENSOR
208
VEHICLE SPEED SENSOR
210

FIG. 10

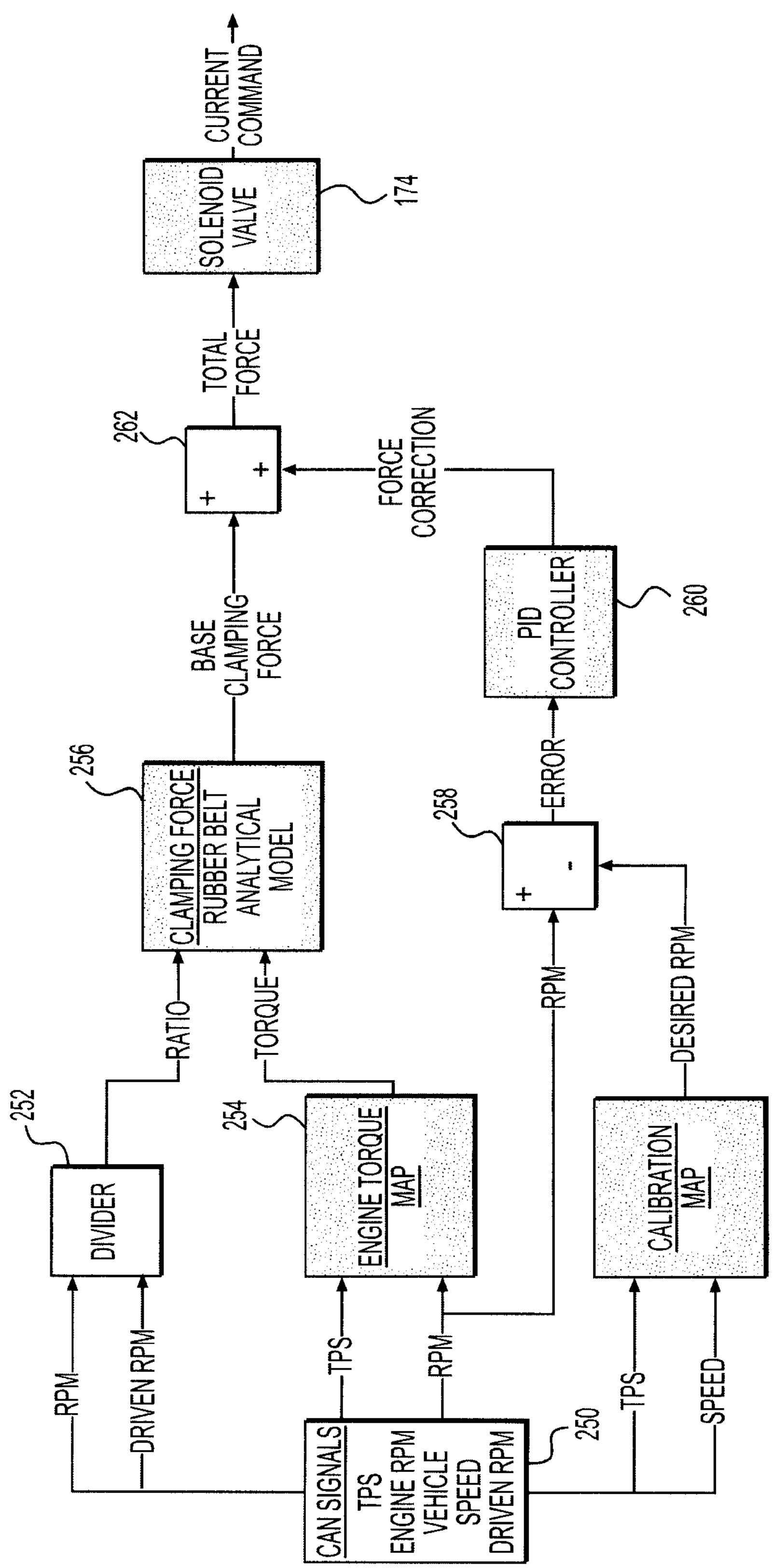
CAN SIGNALS
TPS
ENGINE RPM
VEHICLE SPEED
DRIVEN RPM
250
RPM
DRIVEN RPM
TPS
RPM
TPS
SPEED
DIVIDER
252
ENGINE TORQUE MAP
254
CALIBRATION MAP
RATIO
TORQUE
RPM
DESIRED RPM
CLAMPING FORCE
RUBBER BELT ANALYTICAL MODEL
256
+
-
258
ERROR
PID CONTROLLER
260
BASE CLAMPING FORCE
FORCE CORRECTION
+
+
262
TOTAL FORCE
SOLENOID VALVE
174
CURRENT COMMAND

FIG. 11

		CALIBRATION MAP									
		SPEED (km/h)									
		0	20	40	60	80	100	120	140	160	180
TPS (%)	0	1800	1800	1850	1900	2800	3500	4000	4000	4000	4000
	5	1800	2000	1900	1900	2800	3500	4500	4000	4500	4500
	7	1800	2200	2400	2800	3200	4000	4800	4500	5000	5000
	10	1800	2500	2600	3600	4200	4800	5200	5000	5500	5500
	15	1800	2500	2800	4000	4300	5200	5600	5500	6000	6500
	20	1800	2500	3500	4500	4500	5500	6500	6500	6500	7000
	25	1800	3500	4000	5000	5500	6000	6800	7200	7200	7200
	40	1800	4000	4500	6500	6800	6500	7000	7500	7500	7500
	60	1800	4500	5000	7000	7200	7500	7500	7800	7600	7700
	100	1800	5000	5500	7700	7800	7800	7800	7800	7800	7800

FIG. 12A

		CALIBRATION MAP									
		SPEED (km/h)									
		0	20	40	60	80	100	120	140	160	180
TPS (%)	0	2500	4000	4000	4000	4000	4000	4000	4000	4000	4000
	5	2500	3600	4000	4300	4500	4500	4500	4500	4500	4500
	7	2800	3600	4000	4500	4800	5000	5000	5000	5000	5000
	10	3300	4000	4300	4700	5000	5500	5500	5500	5500	5500
	15	4000	4500	4500	4900	5200	6000	6500	6500	6500	6000
	20	4500	5200	5500	5500	5800	6500	7200	7200	7200	6500
	25	5200	5800	6200	6000	6800	7200	7600	7600	7600	7000
	40	5500	6200	6800	6500	7400	7600	7750	7750	7750	7500
	60	6500	6800	7200	7200	7600	7700	7800	7800	7800	7800
	100	7000	7500	7600	7800	7800	7800	7800	7800	7800	7800

FIG. 12B

ENGINE TORQUE MAP

TPS (%)	ENGINE RPM 1500	2000	3000	3500	4000	4500	5000	6000	7000	8000
0	0.0	5.0	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0
5	5.0	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0
7.5	10.0	15.0	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0
10	20.0	25.0	30.0	35.0	40.0	45.0	50.0	55.0	60.0	65.0
12.5	25.0	30.0	35.0	40.0	45.0	50.0	55.0	60.0	65.0	70.0
15	30.0	35.0	40.0	45.0	50.0	55.0	60.0	65.0	70.0	75.0
20	40.0	45.0	50.0	55.0	60.0	65.0	70.0	75.0	80.0	85.0
30	50.0	55.0	60.0	65.0	70.0	75.0	80.0	85.0	90.0	95.0
40	60.0	65.0	70.0	75.0	80.0	85.0	90.0	95.0	100.0	105.0
60	80.0	85.0	90.0	95.0	100.0	105.0	110.0	115.0	120.0	125.0
100	100.0	105.0	110.0	115.0	120.0	125.0	130.0	135.0	140.0	145.0

FIG. 13

CLAMPING FORCE MAP

RATIO	TORQUE (Nm) 0	10	20	30	40	50	60	70	80	90	100
0.8	500	800	1100	1400	1700	2000	2300	2600	2900	3200	3500
1	600	900	1200	1500	1800	2100	2400	2700	3000	3300	3600
1.3	700	1000	1300	1600	1900	2200	2500	2800	3100	3400	3700
1.5	800	1100	1400	1700	2000	2300	2600	2900	3200	3500	3800
1.7	900	1200	1500	1800	2100	2400	2700	3000	3300	3600	3900
2	1000	1300	1600	1900	2200	2500	2800	3100	3400	3700	4000
2.25	1100	1400	1700	2000	2300	2600	2900	3200	3500	3800	4100
2.5	1200	1500	1800	2100	2400	2700	3000	3300	3600	3900	4200
3	1300	1600	1900	2200	2500	2800	3100	3400	3700	4000	4300
3.5	1400	1700	2000	2300	2600	2900	3200	3500	3800	4100	4400
4	1500	1800	2100	2400	2700	3000	3300	3600	3900	4200	4500

FIG. 14

DRIVING PULLEY FOR A CONTINUOUSLY VARIABLE TRANSMISSION

FIELD OF THE INVENTION

The present invention relates to a driving pulley for a continuously variable transmission.

BACKGROUND OF THE INVENTION

Conventional snowmobile drive trains incorporate a continuously variable transmission (CVT) having a driving pulley that is operatively coupled to the engine crankshaft and a driven pulley coupled to a driven shaft. The driving pulley acts as a clutch and includes a centrifugally actuated adjusting mechanism through which the drive ratio of the CVT is varied progressively as a function of the engine speed and the output torque at the driven pulley. Typically, the driven shaft is a transverse jackshaft which drives the input member of a chain and sprocket reduction drive. The output of reduction drive is coupled to one end of the axle on which are located the drive track drive sprocket wheels.

Although a centrifugal CVT provides many advantages, the fact that the drive ratio of the CVT is directly related to the engine speed causes some disadvantages. One such disadvantage is that the calibration of the driving pulley is always linked with the maximum power output of the engine. Although this results in great acceleration characteristics for the snowmobile, when the snowmobile operates at cruising speeds it results in the engine operating at a greater speed than necessary, high fuel consumption, high noise levels, and a lot of vibrations being transmitted to the riders of the snowmobile.

Therefore, there is a need for a CVT having a drive ratio which is not directly related to the engine speed.

SUMMARY OF THE INVENTION

It is an object of the present invention to ameliorate at least some of the inconveniences present in the prior art.

It is also an object of the present invention to provide a hydraulically actuated driving pulley for a CVT.

By hydraulically controlling the position of the movable sheave relative to the fixed sheave of the driving pulley, a drive ratio of the CVT can be controlled as desired.

In one aspect, the invention provides a driving pulley for a CVT having a shaft, a fixed sheave mounted on the shaft, a first sleeve disposed around the shaft, the first sleeve being operatively connected to the shaft for rotation therewith and being axially movable relative to the shaft, a movable sheave mounted on the first sleeve for rotation and axial movement therewith, a second sleeve disposed around the first sleeve and being connected to the shaft for rotation therewith, a spring biasing the movable sheave away from the fixed sheave, and a CVT chamber. The fixed and movable sheaves are adapted to receive a belt therebetween. The CVT chamber has at least one opening adapted for fluidly communicating the CVT chamber with a hydraulic fluid reservoir. Hydraulic pressure in the CVT chamber biases the movable sheave toward the fixed sheave. The CVT chamber has an annular cross-section. The CVT chamber also has an inner wall being formed by the first sleeve, an outer wall being formed by the second sleeve, an outer end, and an inner end being formed by the movable sheave.

In a further aspect, an annular cover connects the second sleeve to the shaft. The outer end of the CVT chamber is formed by the annular cover.

In an additional aspect, the spring is a helical spring disposed inside the CVT chamber.

In a further aspect, the spring has a first end retained by the first sleeve and a second end retained by the second sleeve.

In an additional aspect, the first end of the spring is disposed near the outer end of the CVT chamber and the second end of the spring is disposed near the inner end of the CVT chamber.

In a further aspect, a portion of the movable sheave is disposed around the second sleeve.

In an additional aspect, at least one seal is disposed between the portion of the movable sheave and the second sleeve.

In a further aspect, a third sleeve is connected inside the portion of the movable sheave. The inner end of the CVT chamber is formed by the third sleeve.

In a further aspect, a fastener is disposed inside the shaft for connecting the driving pulley to a driving shaft of an engine.

In an additional aspect, the at least one opening of the CVT chamber is formed in the first sleeve. The fastener has a passage defined therein. The shaft has a passage defined therein. Hydraulic fluid from the hydraulic fluid reservoir flows sequentially through the passage in the fastener, through the passage in the shaft, through the at least one opening, and in the CVT chamber.

In a further aspect, the passage in the fastener has an axial portion and at least one radially extending outlet. A plane passing through a center of a belt disposed between the fixed and movable sheaves intersects the axial portion of the passage in the fastener. The plane is disposed between at least a portion of the fixed sheave and the at least one radially extending outlet.

In an additional aspect, at least one first axial groove is formed in an outer surface of the shaft. At least one second axial groove is formed in an inner surface of the first sleeve. At least one ball bearing is disposed in the at least one first and the at least one second axial grooves for transmitting torque from the shaft to the first sleeve.

For purposes of this application, the terms related to spatial orientation such as forwardly, rearwardly, left and right, are as they would normally be understood by a driver of a vehicle sitting thereon in a normal driving position.

Embodiments of the present invention each have at least one of the above-mentioned objects and/or aspects, but do not necessarily have all of them. It should be understood that some aspects of the present invention that have resulted from attempting to attain the above-mentioned objects may not satisfy these objects and/or may satisfy other objects not specifically recited herein.

Additional and/or alternative features, aspects, and advantages of embodiments of the present invention will become apparent from the following description, the accompanying drawings, and the appended claims.

BRIEF DESCRIPTION OF THE DRAWINGS

For a better understanding of the present invention, as well as other aspects and further features thereof, reference is made to the following description which is to be used in conjunction with the accompanying drawings, where:

FIG. **1** is a right side elevation view of a snowmobile;

FIG. **2** is a perspective view, taken from a front, left side, of a powertrain of the snowmobile of FIG. **1**;

FIG. **3**A is a cross-sectional view of a driving pulley of a CVT of the powertrain of FIG. **2**;

FIG. **3**B is an exploded view of the driving pulley of FIG. **3**A;

FIG. **4** is an elevation view of a portion of the casing of an engine of the powertrain of FIG. **2** showing elements of a hydraulic system of the CVT;

FIG. **5** is a cross-sectional view of the portion of the casing of FIG. **4** taken through line **5-5** of FIG. **4**;

FIG. **6** is a perspective view of the portion of the casing of FIG. **4** with an inner reservoir cover mounted to the casing;

FIG. **7** is a cross-sectional view of the portion of the casing of FIG. **4** taken through line **7-7** of FIG. **4** with the inner reservoir cover and an outer reservoir cover mounted to the casing;

FIG. **8** is a cross-sectional view of the portion of the casing of FIG. **4** taken through line **8-8** of FIG. **6**;

FIG. **9** is a diagram of the hydraulic system for the CVT;

FIG. **10** is a schematic representation of elements of an electronic system of the snowmobile of FIG. **1**;

FIG. **11** is a flow chart illustrating a method of controlling the CVT;

FIG. **12**A is an example of a calibration map used in the method of controlling the CVT;

FIG. **12**B is an example of another calibration map used in the method of controlling the CVT;

FIG. **13** is an example of an engine torque map used in the method of controlling the CVT; and

FIG. **14** is an example of a clamping force map used in the method of controlling the CVT.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The present invention will be described with respect to a snowmobile. However, it is contemplated that the invention could be used in other vehicles, such as, but not limited to, a motorcycle, a three-wheel vehicle and an all-terrain vehicle (ATV).

Turning now to FIG. **1**, a snowmobile **10** includes a forward end **12** and a rearward end **14** which are defined consistently with a forward travel direction of the vehicle. The snowmobile **10** includes a frame **16** which normally includes a tunnel **18**, an engine cradle portion **20** and a front suspension assembly portion **22**. The tunnel **18** generally consists of sheet metal bent in an inverted U-shape which extends rearwardly along the longitudinal axis **61** of the snowmobile **10** and is connected at the front to the engine cradle portion **20**. An engine **24**, which is schematically illustrated in FIG. **1**, is carried by the engine cradle portion **20** of the frame **16**. The engine **24** has an engine casing **25** (FIG. **2**). The engine casing **25** consists of various parts fastened or otherwise connected to each other. A steering assembly is provided, in which two skis **26** are positioned at the forward end **12** of the snowmobile **10** and are attached to the front suspension assembly portion **22** of the frame **16** through a front suspension assembly **28**. The front suspension assembly **28** includes ski legs **30**, supporting arms **32** and ball joints (not shown) for operatively connecting the respective skis **26** to a steering column **34**. A steering device such as a handlebar **36**, positioned forward of a rider, is attached to the upper end of the steering column **34** to allow the rider to rotate the ski legs **30** and thus the skis **26**, in order to steer the snowmobile **10**.

An endless drive track **65** is positioned at the rear end **14** of the snowmobile **10**. The drive track **65** is disposed generally under the tunnel **18**, and is operatively connected to the engine **24** through CVT **40** illustrated schematically by broken lines and which will be described in greater detail below. The endless drive track **65** is driven to run about a rear suspension assembly **42** for propulsion of the snowmobile **10**. The rear suspension assembly **42** includes a pair of slide rails **44** in sliding contact with the endless drive track **65**. The rear suspension assembly **42** also includes one or more shock absorbers **46** which may further include coil springs (not shown) surrounding the shock absorbers **46**. Suspension arms **48** and **50** are provided to attach the slide rails **44** to the frame **16**. One or more idler wheels **52** are also provided in the rear suspension assembly **42**.

At the front end **12** of the snowmobile **10**, fairings **54** enclose the engine **24** and the CVT **40**, thereby providing an external shell that protects the engine **24** and the CVT **40**, and can also be decorated to make the snowmobile **10** more aesthetically pleasing. The fairings **54** include a hood and one or more side panels which can be opened to allow access to the engine **24** and the CVT **40** when this is required, for example, for inspection or maintenance of the engine **24** and/or the CVT **40**. In the particular snowmobile **10** shown in FIG. **1**, the side panels can be opened along a vertical axis to swing away from the snowmobile **10**. A windshield **56** is connected to the fairings **54** near the front end **12** of the snowmobile **10** or alternatively directly to the handlebar **36**. The windshield **56** acts as a wind screen to lessen the force of the air on the rider while the snowmobile **10** is moving.

The engine **24** is an internal combustion engine that is supported on the frame **16** and is located at the engine cradle portion **20**. The internal construction of the engine **24** may be of any known type and can operate on the two-stroke or four-stroke principle. The engine **24** drives a crankshaft **57** (FIG. **4**) that rotates about a horizontally disposed axis that extends generally transversely to the longitudinal axis **61** of the snowmobile **10**. The crankshaft **57** drives the CVT **40** for transmitting torque to the endless drive track **65** for propulsion of the snowmobile **10** as described in greater detail below.

A straddle-type seat **58** is positioned atop the frame **16**. A rear portion of the seat **58** may include a storage compartment or can be used to accommodate a passenger seat. Two footrests **60** are positioned on opposite sides of the snowmobile **10** below the seat **58** to accommodate the driver's feet.

FIG. **2** illustrates schematically a powertrain **75** of the snowmobile **10**. The powertrain **75** includes the engine **24**, the CVT **40** and a fixed ratio reduction drive **78**. A throttle body **94** having a throttle valve **96** therein is connected to air intake ports of the engine **24** to control the flow of air to the engine **24**. It is contemplated that the throttle body **94** could be replaced by a carburetor. The CVT **40** includes a driving pulley **80** coupled, directly or indirectly, to rotate with the crankshaft **57** of the engine **24** and a driven pulley **88** coupled to one end of a transversely mounted jackshaft **92** which is supported on the frame **16** through bearings. As illustrated, the transversely mounted jackshaft **92** traverses the width of the engine **24**. The opposite end of the transversely mounted jackshaft **92** is connected to the input member of the reduction drive **78** and the output member of the reduction drive **78** is connected to a drive axle **90** carrying sprocket wheels (not shown) that form a driving connection with the drive track **65**.

The driving pulley **80** of the CVT **40** includes a pair of opposed frustoconical belt drive sheaves **82** and **84** between which the drive belt **86** is located. The drive belt is preferably made of rubber. The driving pulley **80** will be described in greater detail below. The driven pulley **88** includes a pair of frustoconical belt drive sheaves **87** and **89** between which the drive belt **86** is located. The driving pulley **80** engages the drive belt **86**. The torque being transmitted to the driven pulley **88** provides the necessary clamping force on the belt **86** through its torque sensitive mechanical device in order to efficiently transfer torque to the further powertrain components. The effective diameters of the driving pulley **80** and the driven pulley **88** are the result of the equilibrium of forces on the drive belt **86** from the hydraulic system of the driving pulley **80** and the torque sensitive mechanism of the driven pulley **88**.

In this particular example, the driving pulley **80** rotates at the same speed as the crankshaft **57** of the engine **24** whereas the speed of rotation of the transverse jackshaft **92** is determined in accordance with the instantaneous ratio of the CVT **40**, and the drive axle **90** rotates at a lower speed than the transverse jackshaft **92** because of the action of the reduction drive **78**. Typically, the input member of the reduction drive **78** consists of a small sprocket connected to the transverse jackshaft **92** and coupled to drive an output member consisting of a larger sprocket connected to the drive axle **90** through a driving chain, all enclosed within the housing of the reduction drive **78**.

It is contemplated that the driving pulley **80** could be coupled to an engine shaft other than the crankshaft **57**, such as an output shaft, a counterbalance shaft, or a power take-off shaft driven by and extending from the engine **24**. The shaft driving the driving pulley **80** is therefore generally referred to as the driving shaft. Although the present embodiment is being described with the crankshaft **57** being the driving shaft, it should be understood that other shafts are contemplated. Similarly, it is contemplated that the driven pulley **88** could be coupled to a shaft other than the transverse jackshaft **92**, such as directly to the drive axle **90** or any other shaft operatively connected to the ground engaging element of the vehicle (i.e. the drive track **65** in the case of the snowmobile **10**). The shaft driven by the driven pulley **88** is therefore generally referred to as the driven shaft. Although the present embodiment is being described with the transverse jackshaft **92** being the driven shaft, it should be understood that other shafts are contemplated.

Turning now to FIGS. **3**A and **3**B, the driving pulley **80** will be described in more detail. As discussed above, the driving pulley **80** includes a pair of opposed frustoconical belt drive sheaves **82** and **84**. Both sheaves **82** and **84** rotate together with the crankshaft **57**. The sheave **82** is fixed in an axial direction of the crankshaft **57**, and is therefore referred to as the fixed sheave **82**. The sheave **84** can move toward or away from the fixed sheave **82** in the axial direction of the crankshaft **57** in order to change the drive ratio of the CVT **40**, and is therefore referred to as the movable sheave **84**. As can be seen in FIG. **2**, the fixed sheave **82** is disposed between the movable sheave **84** and the engine **24**, however it is contemplated that the movable sheave **84** could be disposed between the fixed sheave **82** and the engine **24**.

The fixed sheave **82** is mounted on a shaft **100**. A portion **101** of the shaft **100** is taper-fitted on the end of the crankshaft **57** such that the shaft **100** and the fixed sheave **82** rotate with the crankshaft **57**. It is contemplated that the shaft **100** could be connected to the crankshaft **57** in other known manners. For example, the shaft **100** could engage the crankshaft **57** via splines. A bolt **102** inserted inside the shaft **100** is screwed inside the end of the crankshaft **57**, thus retaining the shaft **100**, and therefore the fixed sheave **82**, on the crankshaft **57**. A sleeve **104** is disposed around the shaft **100**. Ball bearings **103** are disposed in axial grooves **105**, **106** in the outer surface of the shaft **100** and the inner surface of the sleeve **104** respectively. The ball bearings **103** transfer torque from the shaft **100** to the sleeve **104** such that the sleeve **104** rotates with the shaft **100** while permitting axial movement of the sleeve **104** relative to the shaft **100**. Retaining rings **127** disposed on the shaft **100** limit the movement of the ball bearings **103** inside the grooves **105**, **106**. The movable sheave **84** is mounted on the sleeve **104** such that the movable sheave **84** rotates and moves axially with the sleeve **104**, and therefore rotates with the shaft **100** and the crankshaft **57**. A sleeve **107** is press-fit inside the movable sheave **84**. It is contemplated that the sleeve **107** could be omitted.

An annular cover **108** is retained between the end of the shaft **100** and a flanged head of a bolt **109** so as to rotate with the shaft **100**. The bolt **109** is screwed inside the end of the shaft **100**. A cap **111** is clipped in the end of the bolt **109**. A sleeve **113** is connected to the annular cover **108** by screws **115** and is received axially between portions of the movable sheave **84** and of the sleeve **104**.

A CVT chamber **110** is defined between the annular cover **108** and the sleeves **104**, **107**, and **113**. The CVT chamber **110** has an annular cross-section. An inner wall of the CVT chamber **110** is formed by the sleeve **104**, an outer wall of the CVT chamber **110** is formed by the sleeve **113**, an outer end of the CVT chamber **110** is formed by the annular cover **108**, and an inner end of the CVT chamber is formed by the sleeve **107** (or the movable sheave **84** should the sleeve **107** be omitted). A helical spring **112** is disposed inside the CVT chamber **110**. One end of the spring **112** abuts a ring **117** abutting the sleeve **113** which is axially fixed relative to the crankshaft **57**. The other end of the spring **112** abuts a ring **119** which abuts a clip **129** connected to the sleeve **104** which is axially movable relative to the crankshaft **57**. This arrangement of the spring **112** causes the spring **112** to bias the movable sheave **84** away from the fixed sheave **82**.

As will be explained in greater detail below, hydraulic pressure created by hydraulic fluid supplied to the CVT chamber **110** biases the movable sheave **84** toward the fixed sheave **82** in order to change the drive ratio of the CVT **40**. As can be seen in FIG. **5**, the crankshaft **57** has an axial passage **114** extending axially therein and two inlet passages **116** extending radially from the axial passage **114** to the outer surface of the crankshaft **57**. Although shown as extending perpendicularly and radially from the axial passage **114**, it is contemplated that the inlet passages **116** could extend radially at some other angle from the axial passage **114**. It is also contemplated that more than two inlet passages **116** or only one inlet passage **116** could be provided. As explained below, a pump **118** (FIG. **6**) supplies hydraulic fluid, such as oil for example, to the axial passage **114** of the crankshaft **57** via the inlet passages **116**. Returning now to FIG. **3**A, from the axial passage **114**, the hydraulic fluid flows in a passage **121** defined in the bolt **102**. The passage **121** has an axial portion and multiple radially extending outlets. As can be seen in FIG. **3**A, when the movable sheave **84** is biased toward the fixed sheave **82** (as illustrated by the movable sheave **84** shown schematically in dotted lines in this figure), a plane **85** passing through a center of the belt **86** intersects the passages **121** and **114** and is disposed laterally between the inlet passages **116** of the crankshaft **57** and the radially extending outlets of the passage **121** of the bolt **102**. The hydraulic fluid then flows through passages **123** in the shaft **100**, through grooves **105**, **106** and through passages **120** in the sleeve **104** into the CVT chamber **110**. As the hydraulic pressure increases inside the CVT chamber **110**, the movable sheave **84** moves axially toward the fixed sheave **82**. When the hydraulic pressure inside the CVT chamber **110** is reduced, as will be described below, the bias of the spring **112** causes the movable sheave **84** to move axially away from the fixed sheave **82** and the hydraulic fluid flows out of the CVT chamber **110** in the direction opposite to what has been described above.

Seals **122** disposed between the sleeve **113** and the sleeve **107**, seals **124** disposed between the shaft **100** and the sleeve **104**, and various O-rings **125** prevent hydraulic fluid from leaking out of the driving pulley **80**.

By having the hydraulic fluid supplied to the CVT chamber **110** via a driving shaft extending from the engine **24**, the belt **86** can easily be removed from the pulleys **80**, **88** for maintenance or replacement since no portion of the hydraulic system of the CVT **40** extends on a side of the CVT **40** opposite the side on which the engine **24** is disposed (i.e. the belt **86** is removed over the movable sheave **84** from a side of the driving pulley **84** opposite the side from which hydraulic fluid enters the driving pulley **84**).

Turning now to FIGS. **4** to **8**, the hydraulic system supplying hydraulic fluid to the CVT chamber **110** will be described. Although, the system will be described with respect to these figures, for simplicity of understanding, reference can be made to FIG. **9** which provides a diagrammatic representation of the hydraulic system.

The hydraulic system has a first reservoir **126** for holding the hydraulic fluid. The first reservoir **126** is formed between the engine casing **25** and a cover **128** (FIG. **7**) sealingly connected to a protruding lip **130** of the engine casing **25**. The portion of the engine casing **25** forming the first reservoir **126** is fastened to other portions of the engine casing **25**. However it is contemplated that it could be integrally formed with another portion of the engine casing **25**, such as the crankcase for example. From the first reservoir **126**, the hydraulic fluid flows through a filter **132** located near a bottom of the first reservoir **126** and then flows in a passage **134**. From the passage **134**, the hydraulic fluid enters the pump **118** and flows out of the pump **118** into a second reservoir (or canal) **136**. The second reservoir **136** is formed between the engine casing **25** and a cover **138** (best seen in FIG. **6**) sealingly connected to a protruding lip **140** of the engine casing **25**. As best seen in FIG. **4**, the second reservoir **136** surrounds the crankshaft **57**, and the first reservoir **126** surrounds the second reservoir **136**. Seals **137** are disposed around the crankshaft **57** on either side of the inlet passages **116** (see FIG. **5**). The pump **118** is preferably a gerotor pump driven by the engine **24**. As seen in FIG. **6**, the gerotor pump consists of an inner rotor **142** disposed off-center from an outer rotor **144**, with both rotors **142**, **144** rotating when the pump **118** is in operation. It is contemplated that other types of pumps could be used. It is also contemplated that the pump could be driven separately from the engine **24**, such as by an electric motor for example. While the pump **118** is operating, the hydraulic pressure inside the second reservoir **136** is normally greater than in the first reservoir **126**. A pressure release valve **146** is disposed in a passage in the protruding lip **140** so as to fluidly communicate the second reservoir **136** with the first reservoir **126** should the hydraulic pressure inside the second reservoir **136** become too high. From the second reservoir **136**, the hydraulic fluid flows to the inlet passages **116** of the crankshaft **57** and then to the CVT chamber **110** as described above.

As best seen in FIG. **7**, the hydraulic system is provided with a piloted proportional pressure relief valve **148**. It is contemplated that a non-piloted valve could be provided instead of the piloted proportional pressure relief valve **148**. The piloted proportional pressure relief valve **148** controls fluid communication between the second reservoir **136** and the first reservoir **126** so as to control a hydraulic pressure in the second reservoir **136**. By controlling the hydraulic pressure in the second reservoir **136**, the hydraulic pressure in the CVT chamber **110** is also controlled, which in turn controls the position of the movable sheave **84** with respect to the fixed sheave **82**, and therefore controls the drive ratio of the CVT **40**.

The piloted proportional pressure relief valve **148** has a bell-shaped upper end **150** disposed in the second reservoir **136** near an outlet of the pump **118**. A lower end **152** of the piloted proportional pressure relief valve **148** closes and opens a passage **154** from the second reservoir. A piloted proportional pressure relief valve chamber **156** is disposed adjacent the lower end **152** of the piloted proportional pressure relief valve **148**. The piloted proportional pressure relief valve chamber **156** contains hydraulic fluid. The hydraulic pressure in the piloted proportional pressure relief valve chamber **156** biases the piloted proportional pressure relief valve **148** upwardly toward its closed position (i.e. the position shown in FIG. **7**, with the lower end **152** of the piloted proportional pressure relief valve closing the passage **154** completely). The amount of hydraulic pressure in the piloted proportional pressure relief valve chamber **156**, and therefore the amount of upward bias on the piloted proportional pressure relief valve, can be controlled as will be described below. A spring **158** is disposed in the piloted proportional pressure relief valve chamber **156** between the lower end **152** of the piloted proportional pressure relief valve and the upper end of a threaded plug **160**. The spring **158** also biases the piloted proportional pressure relief valve **148** upwardly toward its closed position. By screwing and unscrewing the threaded plug **160**, a degree of preloading of the spring **158** can be adjusted which in turn controls the amount of bias provided by the spring **158**. The hydraulic pressure on the bell-shaped upper end **150** biases the piloted proportional pressure relief valve **148** downwardly toward an opened position (i.e. a position where the lower end **152** of the piloted proportional pressure relief valve **148** does not close the passage **154** completely). It should be understood that the piloted proportional pressure relief valve **148** has multiple opened positions each providing a different degree of opening of the passage **154**. When the downward force on the piloted proportional pressure relief valve **148** due to the hydraulic pressure acting on the upper end **150** exceeds the upward force on the piloted proportional pressure relief valve **148** due to the hydraulic pressure acting on the lower end **152** and the bias of the spring **158**, the piloted proportional pressure relief valve **148** moves downwardly to an opened position.

When the piloted proportional pressure relief valve **148** is in an opened position, hydraulic fluid flows through the passage **154** from the second reservoir **136**, to a chamber **162** disposed between the ends **150**, **152** of the piloted proportional pressure relief valve **148**. From the chamber **162**, the hydraulic fluid flows into a return passage **164** (best seen in FIG. **6**) to the first reservoir **126**. The return passage **164** is formed between the cover **138** and a metal gasket **165** (best seen in FIG. **8**) disposed between the cover **138** and the protruding lip **140**. The outlet **166** of the return passage **164** is located near the bottom of the first reservoir **126** such that the outlet **166** is disposed below a level of hydraulic fluid in the first reservoir **126**, thus reducing the likelihood of air bubbles being formed by the hydraulic fluid flowing into the first reservoir **126** from the return passage **164**. Therefore, as the degree of opening of the piloted proportional pressure relief valve **148** is increased, the hydraulic pressure in the second reservoir **136** is reduced, which reduces the hydraulic pressure in the CVT chamber **110**, which in turn causes the movable sheave **84** to move away from the fixed sheave **82** due to the bias of the spring **112**. As the degree of opening of the piloted proportional pressure relief valve **148** is decreased, the hydraulic pressure in the second reservoir **136** is increased, which increases the hydraulic pressure in the CVT chamber **110**, which in turn causes the movable sheave **84** to move toward the fixed sheave **82**. Thus, by controlling a degree of opening of the piloted proportional pressure relief valve **148** as described below, the position of the movable sheave **84** with respect to the fixed sheave **82**, and therefore the drive ratio of the CVT **40**, can be controlled.

The piloted proportional pressure relief valve chamber **156** fluidly communicates with a piloted proportional pressure relief valve passage **168** (best seen in FIG. **6**). The piloted proportional pressure relief valve passage **168** is formed between the cover **138** and the metal gasket **165**. The piloted proportional pressure relief valve passage **168** extends upwardly to a pilot valve chamber **170** (FIG. **8**). As seen in FIG. **8**, an opening **172** in the metal gasket **165** communicates the piloted proportional pressure relief valve passage **168** with the second reservoir **136** such that hydraulic fluid can be supplied from the second reservoir **136** to the piloted proportional pressure relief valve chamber **156** via the piloted proportional pressure relief valve passage **168**. An electronically controlled pilot valve in the form of a solenoid **174** is disposed adjacent to the pilot valve chamber **170**. It is contemplated that other types of electronically controlled pilot valve could be used. The solenoid **174** is held in a holder **180**. A passage **176** in the solenoid fluidly communicates the pilot valve chamber **170** with the first reservoir **126**. The solenoid **174** and passage **176** together form an electronically controlled pilot valve. The solenoid **174** modulates the forces applied on a movable end **178** thereof (shown in FIG. **8**) in response to a signal received from a control unit **200** (FIG. **10**) as described below. The pressure in the pilot valve chamber **170** is then proportional to the force exerted by the solenoid **174** on its movable end **178**. The movable end **178** modulates a degree of opening the passage **176** to compensate for the flow variation coming from the reservoir **136** through the opening **172**. As explained above, decreasing the hydraulic pressure in the piloted proportional pressure relief valve chamber **156** reduces the upward bias on the piloted proportional pressure relief valve **148** which in turn reduces the hydraulic pressure in the CVT chamber **110**, thus causing the sheaves **82**,**84** to move away from each other. When the end **178** of the solenoid **174** reduces the degree of opening of the passage **176**, hydraulic fluid flowing in the opening **172** from the second reservoir **136** increases the hydraulic pressure in the piloted proportional pressure relief valve chamber **156**. As explained above, increasing the hydraulic pressure in the piloted proportional pressure relief valve chamber **156** increases the upward bias on the piloted proportional pressure relief valve **148** which in turn increases the hydraulic pressure in the CVT chamber **110**, thus causing the sheaves **82**, **84** to move toward each other. Thus, controlling an opening and closing cycle of the end **178** of the solenoid **174**, controls an opening and closing cycle of the passage **176**, which in turns controls the position of the movable sheave **84** with respect to the fixed sheave **82**, and therefore the drive ratio of the CVT **40**. The opening **172** has a smaller cross-sectional area than the cross-sectional area of the passage **176** which causes a drop in pressure between the reservoir **136** and the pilot valve chamber **170**. In a preferred embodiment, the opening **172** has a circular cross-section having a 0.8 mm diameter, and the passage **176** has a circular cross-section having a 3 mm diameter.

Turning now to FIG. **10**, elements of an electronic system of the snowmobile **10** used to control the drive ratio of the CVT **40** will be described. The electronic system includes the control unit **200**. The control unit **200** receives signals from a number of sensors (described below), uses these signals to determine a clamping force to be applied to the belt **86**, as described in greater detail below, such as to obtain a desired drive ratio of the CVT **40**. The clamping force is the force applied on either side of the belt **86** by the sheaves **82**, **84** in the axial direction of the crankshaft **57**. Based on the clamping force, the control unit **200** sends a signal to a piloted valve actuator **202** to control an opening and closing cycle of the piloted proportional pressure relief valve **148** in order to obtain a hydraulic pressure in the CVT chamber **110** that will provide the clamping force to be applied. The signal sent from the control unit **200** to the piloted valve actuator **202** is preferably a pulse-width modulated (PWM) signal. In the present embodiment, the piloted valve actuator **202** consists of the solenoid **174** which is used to control the hydraulic pressure in the piloted proportional pressure relief valve chamber **156** as described above. However, it is contemplated that other types or arrangements of piloted valve actuators could be used. For example, the piloted valve actuator **202** could be a solenoid mechanically actuating the valve **148**.

A driving shaft speed sensor **204** senses a speed of rotation of the crankshaft **57** (or other driving shaft associated with the driving pulley **80**) and sends a signal representative of the speed of rotation of the crankshaft **57** to the control unit **200**. A throttle position sensor **208** senses a position of the throttle valve **96** and sends a signal representative of this position to the control unit **200**. The position of the throttle valve **96** is preferable determined as a percentage of opening of the throttle valve **96** (0% being a fully closed position and 100% being a fully opened position), however it is contemplated that the position of the throttle valve **96** could be determined in terms of degrees of opening or any other suitable terms. A vehicle speed sensor **210** senses a speed of the snowmobile **10** and sends a signal representative of this speed to the control unit **200**. The control unit **200** determines the speed of rotation of the driven shaft (i.e. the jackshaft **92**) from the signal received from the speed sensor **210**. It is contemplated that driven shaft speed sensor could be provided to sense a speed of rotation of the driven shaft and send a signal representative of the speed of rotation of the driven shaft to the control unit **200**. The above sensors **204**, **208** and **210** could be of any type suitable for their intended purposes, as would be understood by a person skilled in the art. The signals sent from the sensors **204**, **208** and **210** to the control unit **200** preferably use a Controller-Area Network (CAN) protocol.

Turning now to FIG. **11**, the method by which the control unit **200** determines the clamping force to be applied to the belt **86** by the driving pulley **80**, and from which the control unit **200** determines the signal to be sent to the solenoid **174** (or other piloted valve actuator **202**) will be described. From the signals **250** received from the sensors **204**, **208** and **210**, the control unit **200** determines the current drive ratio of the CVT **40** by running the speed of rotation of the crankshaft **57** and the speed of rotation of the driven shaft through a divider **252** (i.e. drive ratio=driving speed/driven speed). It is contemplated that the control unit **200** could determine the drive ratio of the CVT **40** by using other inputs and methods. For example, the drive ratio of the CVT **40** could be determine by comparing the distance between the sheaves **82**, **84** of the driving pulley **80** to the distance between the sheaves **87**, **89** of the driven pulley **88**. The control unit **200** also determines the engine torque by using the position of the throttle valve **96** and the speed of rotation of the crankshaft **57** together with an engine torque map **254** such as the one shown in FIG. **13**. In FIG. **13**, the position of the throttle valve **96** appears in terms of percentage of opening of the throttle valve **96**. The engine torques given in the table of FIG. **13** are in Newton-meters (Nm). It is contemplated that the control unit **200** could determine the engine torque by using other inputs and methods.

By using the current drive ratio of the CVT **40** and the engine torque determined above, the control unit **200** determines a base clamping force. The determination of the base clamping force is made using an analytical model **256**. FIG. **14** shows a clamping force map which was made based on the analytical model. The base clamping forces given in the table of FIG. **14** are in Newtons (N).

The control unit **200** also determines a desired speed of rotation of the crankshaft **57** by using the position of the throttle valve **96** and the speed of the snowmobile **10** together with a calibration map such as one of the ones shown in FIGS. **12**A and **12**B. The desired speeds of rotation of the crankshaft **57** given in the tables of FIGS. **12**A and **12**B are in rotations per minute (RPM).

In one embodiment, the driver of the snowmobile **10** can select one of two or more driving modes using a manually actuated switch **62** (FIG. **1**), where each driving mode has a corresponding calibration map. The selected driving mode is preferably displayed to the driver on a display cluster (not shown) of the snowmobile **10**. For example, in a snowmobile **10** having two driving modes, the calibration map shown in FIG. **12**A could correspond to a "fuel economy" mode and the calibration map in FIG. **12**B could correspond to a "performance" mode. As their names suggest, the calibration map of FIG. **12**A provides good fuel consumption while the calibration map of FIG. **12**B provides improved vehicle performances compared to the "fuel economy" mode.

It is contemplated that the control unit **200** could determine the desired speeds of rotation of the crankshaft **57** by using other inputs and methods.

The values given in FIGS. **12**A to **14** are for exemplary purposes. It should be understood that these values would vary depending on the vehicle, powertrain, and/or CVT characteristics and the desired performance characteristics of the vehicle. For example, the clamping force values given in the map of FIG. **14** would vary depending on the spring constant of the spring **112**.

The control unit **200** then determines a difference (error) between the current speed of rotation of the crankshaft **57** and the desired speed of rotation of the crankshaft **57** determined above by running these values through a comparator **258**. This difference is then inserted in a proportional-integral-derivative (PID) controller **260** which determines a corrective clamping force. It is contemplated that the control unit **200** could determine the corrective clamping force by using other types of controllers.

The base clamping force and the corrective clamping force determined above are then added using a summer **262** to obtain a total clamping force. The control unit **200** finally sends a signal to the solenoid **174** controlling a pulse-width-modulation duty cycle which modulates the degree of opening of the passage **176** such that a resulting hydraulic pressure in the CVT chamber **110** will cause the movable sheave **84** to apply the total clamping force to the belt **86**, thus controlling the drive ratio of the CVT **40**. The total clamping force is lower than the base clamping force when the desired speed of rotation of the crankshaft **57** is higher than the current speed of rotation of the crankshaft **57**. The total clamping force is higher than the base clamping force when the desired speed of rotation of the crankshaft **57** is lower than the current speed of rotation of the crankshaft **57**.

In the embodiment where the driver of the snowmobile **10** can switch from between the calibration maps of FIGS. **12**A and **12**B, during operation of the snowmobile **10**, switching from the calibration map of FIG. **12**B to the calibration map of FIG. **12**A will generally result in the speed of rotation of the crankshaft **57** to decrease (since the desired speed of rotation of the crankshaft **57** decreases) and in the total clamping force to increase, thus maintaining the speed of the snowmobile **10**.

It is contemplated that the summer **262** could be replaced by a comparator. In such an embodiment, either the inputs to the comparator **258** are inverted or the PID controller **260** has a negative gain.

The calibration map, engine torque map, clamping force map, and the PID controller **260** are preferably set such that once the snowmobile **10** reaches a desired (i.e. constant) speed following an acceleration, the total clamping force can be increased. This allows a speed of rotation of the crankshaft **57** to be reduced while still maintaining the speed of the snowmobile **10** constant. It is contemplated that, depending on the engine configuration, a degree of opening of the throttle valve **96** may have to be increased in order to maintain the speed of the snowmobile **10** constant. This results in improved fuel consumption compared to a snowmobile having a centrifugal CVT.

It is contemplated that the calibration map, engine torque map, clamping force map, and the PID controller **260** could also be set such that as the position of the throttle valve **96** decreases, a rate of reduction of the total clamping force is lower than a rate of reduction of the position of the throttle valve **96** which causes engine braking.

Modifications and improvements to the above-described embodiments of the present invention may become apparent to those skilled in the art. The foregoing description is intended to be exemplary rather than limiting. The scope of the present invention is therefore intended to be limited solely by the scope of the appended claims.

What is claimed is:

1. A driving pulley for a continuously variable transmission comprising:

- a shaft having a passage defined therein;
- a fixed sheave mounted on the shaft;
- a first sleeve disposed around the shaft, the first sleeve being operatively connected to the shaft for rotation therewith, and being axially movable relative to the shaft;
- a movable sheave mounted on the first sleeve for rotation and axial movement therewith, the fixed and movable sheaves being adapted to receive a belt therebetween, the passage of the shaft and at least a portion of the fixed sheave being disposed on opposite sides of a plane passing through a center of the belt;
- a second sleeve disposed around the first sleeve, the second sleeve being connected to the shaft for rotation therewith;
- a spring biasing the movable sheave away from the fixed sheave; and
- a CVT chamber having at least one opening adapted for fluidly communicating the CVT chamber with a hydraulic fluid reservoir,
- hydraulic fluid from the hydraulic fluid reservoir flows sequentially through the shaft from one side of the plane to an other side of the plane where the passage of the shaft is located, through the passage in the shaft and in the CVT chamber,
- hydraulic pressure in the CVT chamber biasing the movable sheave toward the fixed sheave, the CVT chamber having an annular cross-section, and the CVT chamber having:
  - an inner wall being formed by the first sleeve,
  - an outer wall being formed by the second sleeve,
  - an outer end, and
  - an inner end being formed by the movable sheave.

2. The pulley of claim **1**, further comprising an annular cover connecting the second sleeve to the shaft;

wherein the outer end of the CVT chamber is formed by the annular cover.

3. The pulley of claim **1**, wherein the spring is a helical spring disposed inside the CVT chamber.

4. The pulley of claim **3**, wherein the spring has a first end retained by the first sleeve and a second end retained by the second sleeve.

5. The pulley of claim **4**, wherein the first end of the spring is disposed near the outer end of the CVT chamber and the second end of the spring is disposed near the inner end of the CVT chamber.

6. The pulley of claim **1**, wherein a portion of the movable sheave is disposed around the second sleeve.

7. The pulley of claim **6**, further comprising at least one seal disposed between the portion of the movable sheave and the second sleeve.

8. The pulley of claim **6**, further comprising a third sleeve connected inside the portion of the movable sheave;

wherein the inner end of the CVT chamber is formed by the third sleeve.

9. The pulley of claim **1**, further comprising a fastener disposed inside the shaft for connecting the driving pulley to a driving shaft of an engine.

10. The pulley of claim **9**, wherein the at least one opening of the CVT chamber is formed in the first sleeve;

wherein the fastener has a passage defined therein; and

wherein hydraulic fluid from the hydraulic fluid reservoir flows sequentially through the passage in the fastener, through the passage in the shaft, through the at least one opening, and in the CVT chamber.

11. The pulley of claim **10**, wherein the passage in the fastener has an axial portion and at least one radially extending outlet;

wherein the plane intersects the axial portion of the passage in the fastener; and

wherein the plane is disposed between at least a portion of the fixed sheave and the at least one radially extending outlet.

12. The pulley of claim **1**, further comprising:

at least one first axial groove formed in an outer surface of the shaft;

at least one second axial groove formed in an inner surface of the first sleeve; and

at least one ball bearing disposed in the at least one first and the at least one second axial grooves for transmitting torque from the shaft to the first sleeve.

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